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The Theory and Design of a Centrifugal Pump

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THE THEORY AND DESIGN OF A CENTRIFUGAL PUMP

BY

ROSS B. WILSON

THESIS

FOR THE

DEGREE OF BACHELOR OF SCIENCE

IN

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THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

Ross B. Wilson

ENTITLED The Theory and Design of a Centrifugal Pump

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE

DEGREE OF Bachelor of Science in Mechanical Engineering

Alvin Schaller.
Instructor in Charge.

APPROVED: G. A. Goodenough

Acting HEAD OF DEPARTMENT OF Mechanical Engineering

197772

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Contents.

Chapter I.
Introduction.

Although centrifugal pumps have been in use a great many years, it has been only in the last few that very much concerning the theory of their design has been published. One of the latest of these publications is, "Die Zentrifugalpumpen", by Fritz von Neuman. Mr. Alvin Schaller kindly offered to read this German text and take all notes necessary for the design of a pump. It was from these notes that all theory, with the exception of the derivation of the fundamental equation, and methods were obtained.

The problem chosen was a pump of the single-suction type with a three inch discharge and

four inch suction. These dimensions agree with current American practise as does also the discharge which was taken as two hundred and sixty gallons per minute against a head of twenty five feet.

In designing this pump, the work falls naturally into five divisions, which are:

(a) The developement of the relation between peripherial velocty, the angle of discharge, and the angle of the vanes.

(b) The design of the impeller.

(c) The design of the casing.

(d) The design of the shaft.

(e) The design of the details of the pump.

Chapter II.
The Fundamental Equation.

The following derivation of the relation between the peripherial velocity and the head is due to Professor G. A. Goodenough of the University of Illinois.

A particle of water in going through the impeller travels along some path such as AB, in Fig. I. We will assume the water to pass through a tube of cross-section f along AB. If we let γ be the specific weight of the water, the mass of a length ds will be the product of f, γ, and ds divided by g. Denoting this by m, we have

$$m = \frac{f\,ds\,\gamma}{g}$$

We will now make the

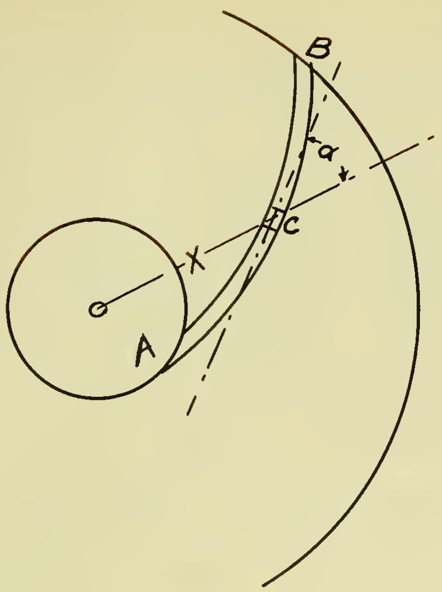


Fig. I.

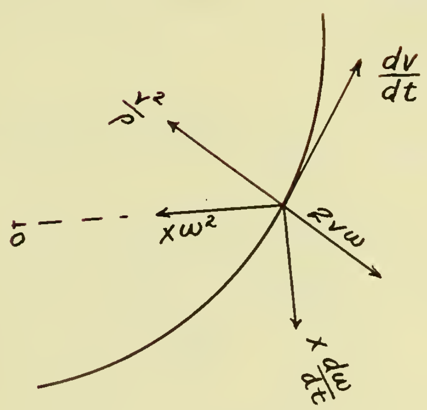


Fig. II.

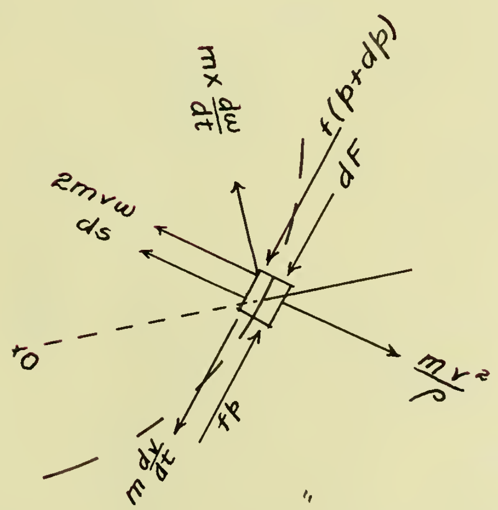


Fig. III.

assumption that this mass is at a point C at a distance x from the center. By Coriolis Law, the acceleration of the mass at C has three components:

(1) The acceleration of the mass with AB at rest.

(2) The acceleration of the point of the rotor coincident with C.

(3) An acceleration normal to the tangent CT.

These accelerations can now be resolved into components parallel and normal to the the paths AB and the line OC.

Let V be the velocity in the curve, ρ be the radius of curvature at C, and ω the angular velocity of the impeller.

The components of (1) then are $\frac{dV}{dt}$ and $\frac{V^2}{\rho}$. The components of (2) are $x\omega^2$ and $x\frac{d\omega}{dt}$. The acceleration (3) is $2V\omega$.

We now have the acc-

elevations replaced by their components as shown in Fig. II.

The next step is to replace these accelerations by the forces producing them reversed. The forces acting are:

(1) Pressure ds between the small mass and the side of the tube.

(2) Friction dF between the water and the tube.

(3) A pressure fp on the cross-section nearest the inner circumference.

(4) A pressure $(p+dp)f$ at a distance ds from the point.

(5) The accelerating forces.

If the accelerating forces are reversed as shown in Fig. III, the system will be in equilibrium and we obtain:

$$f(p+dp) - fp + m\frac{dv}{dt} - mx\omega^2 \cos\alpha + dF = 0$$

Since $m = \frac{\gamma f\,ds}{g}$, we get;

$$f\,dp + \frac{\gamma f\,ds}{g}\frac{dv}{dt} - \frac{\gamma f\,ds}{g}\omega^2 x \cos\alpha + dF = 0$$

Now $V = \frac{ds}{dt}$ and $ds \cos\alpha = dx$.
From which;

$$\int_{p_1}^{p} dp + \gamma \int_{V_1}^{V} \frac{V\,dV}{g} - \frac{\gamma\omega^2}{g}\int_{r_1}^{r} x\,dx + \frac{1}{f}\int dF = 0$$

Integrating, we obtain;

$$(p - p_1) + \frac{\gamma}{2g}(V^2 - V_1^2) + \frac{\gamma\omega^2}{2g}(r^2 - r_1^2) + \frac{1}{f}\int dF = 0$$

We do not definately know $\int dF$, so we assume $\frac{1}{f}\int dF = \mathfrak{S}_r$. Now, since $u_1 = r_1\omega$ and $u = r\omega$, we get.

$$\frac{p - p_1}{\gamma} = \frac{1}{2g}\left(u^2 - u_1^2 + V^2 - V_1^2\right) - \mathfrak{S}_r. \qquad (A)$$

This equation may be interpreted as follows. The change in pressure head is equal to the change in velocity head less the friction head.

Now $V_1^2 = v_1^2 - u_1^2$ and $V^2 = v^2 + u^2 - 2uv\cos\alpha$. Also $\frac{v}{u} = \frac{\sin\beta}{\sin(\beta - \theta)}$.

Substituting the above values for V_1 and V in equation (A), we have.

$$\frac{p - p_1}{\gamma} = \frac{1}{2g}\left(v_1^2 - v^2 + 2uv\cos\alpha\right) - \mathfrak{S}_r$$

Eliminating v by means of the $\frac{v}{u}$ relation, we have;

$$\frac{p-p_1}{\gamma} = \frac{u^2}{2g}\left[\frac{2\sin\beta\cos\alpha}{\sin(\beta-\alpha)} - \frac{\sin^2\beta}{\sin^2(\beta-\alpha)}\right] + \frac{V_1}{2g} - \xi_r$$

We will call this equation (B)

Now at entrance,

$$h_a - k = \frac{v_1^2}{2g} + \frac{p_1}{\gamma} + \xi_1 \qquad (C)$$

Where h_a = atmospheric head, k = total effective lift; ξ_1 = friction head at entrance.

At discharge,

$$\frac{p_2}{\gamma} + \frac{V^2}{2g} = \frac{t^2}{2g} + h + h_a + \xi_d \qquad (D)$$

Where ξ_d is the friction head due to discharge pipe; t is the velocity of the water at the end of the discharge pipe; and h is the lift or head above pump.

Subtracting (C) from (D) and transposing we get,

$$\frac{1}{\gamma}(p_2 - p_1) = h + k + \frac{v_1^2 - V^2}{2g} + \frac{t^2}{2g} + \xi_1 + \xi_d \qquad (E)$$

Comparing with (B), we obtain,

$$\frac{vu\cos\alpha}{g} = h + k + \frac{t^2}{2g} + \xi_1 + \xi_d + \xi_r \qquad (F)$$

Eliminating v by means of the $\frac{v}{u}$ relation and calling $H = h + k$ and $\xi = \xi_1 + \xi_d + \xi_r$, we obtain;

$$\frac{u^2}{g} \quad \frac{\sin\beta\cos\alpha}{\sin(\beta-\alpha)} = H + \frac{t^2}{2g} + \xi$$

The right side of this equation is composed of three parts, the head pumped against or H, the velocity head in the discharge pipe or $\frac{t^2}{2g}$, and the sum of the friction heads. We can group these three heads together as some function of H. This function we will call ηH, η being the reciprocal of the hydraulic efficiency. Substituting ηH in the right hand side and solving for u, we get:

$$u = \sqrt{(\eta g H)(1 - \tan\alpha \cot\beta}.$$

This is the fundamental equation for the relation between the peripherial velocity, the angle of discharge, and the angle of the vanes.

Chapter III
Design of Impeller.

The first step in the design of the impeller is the choice of the notation to be used. In this work we will follow that used by Neumann in his, "Zentrifugalpumpen", from which this theory was obtained.

Let:

z = The number of vanes.

Q' = Total discharge through the impeller.

Q = Discharge of pump.

D_a^a = Outer diameter of impeller.

D_a = Diameter of impeller at a point in the center of a line drawn normal to one vane and passing through the tip of the other.

D_e = An assumed diameter upon which all entrance calculations

are based. D_e corresponds to D_a.

D_s = Diameter of the suction.

d_a = Diameter of the base circle upon which the involute for the blades at exit is constructed.

d_e = Diameter of the base circle at entrance.

β_a^a = Angle of the vanes at the diameter D_a^a.

β_a = Angle of the vanes at the diameter D_a

β_e = Angle of the vanes at the diameter D_e.

α = Angle of discharge.

u_a = Peripherial velocity at the diameter D_a.

u_e = Peripherial velocity at the diameter D_e.

v_a = Velocity relative to the vanes at D_a.

v_e = Velocity relative to the vanes at D_e.

w_a = Absolute velocity at D_a.

w_e = Absolute velocity at D_e.

V_r = The radial component of the relative velocity V_a.

W_s' = The velocity of suction.

b_a = The thickness of the impeller at D_a.

b_e = The thickness of the impeller at D_e.

F_a = The area of discharge at the diameter D_a.

F_e = The area of entrance at the diameter D_e.

η = The reciprocal of the hydraulic efficiency. Assumed in this case to be 1.43.

s_a = Thickness of vanes.

a_a = The perpendicular width of the channel at exit.

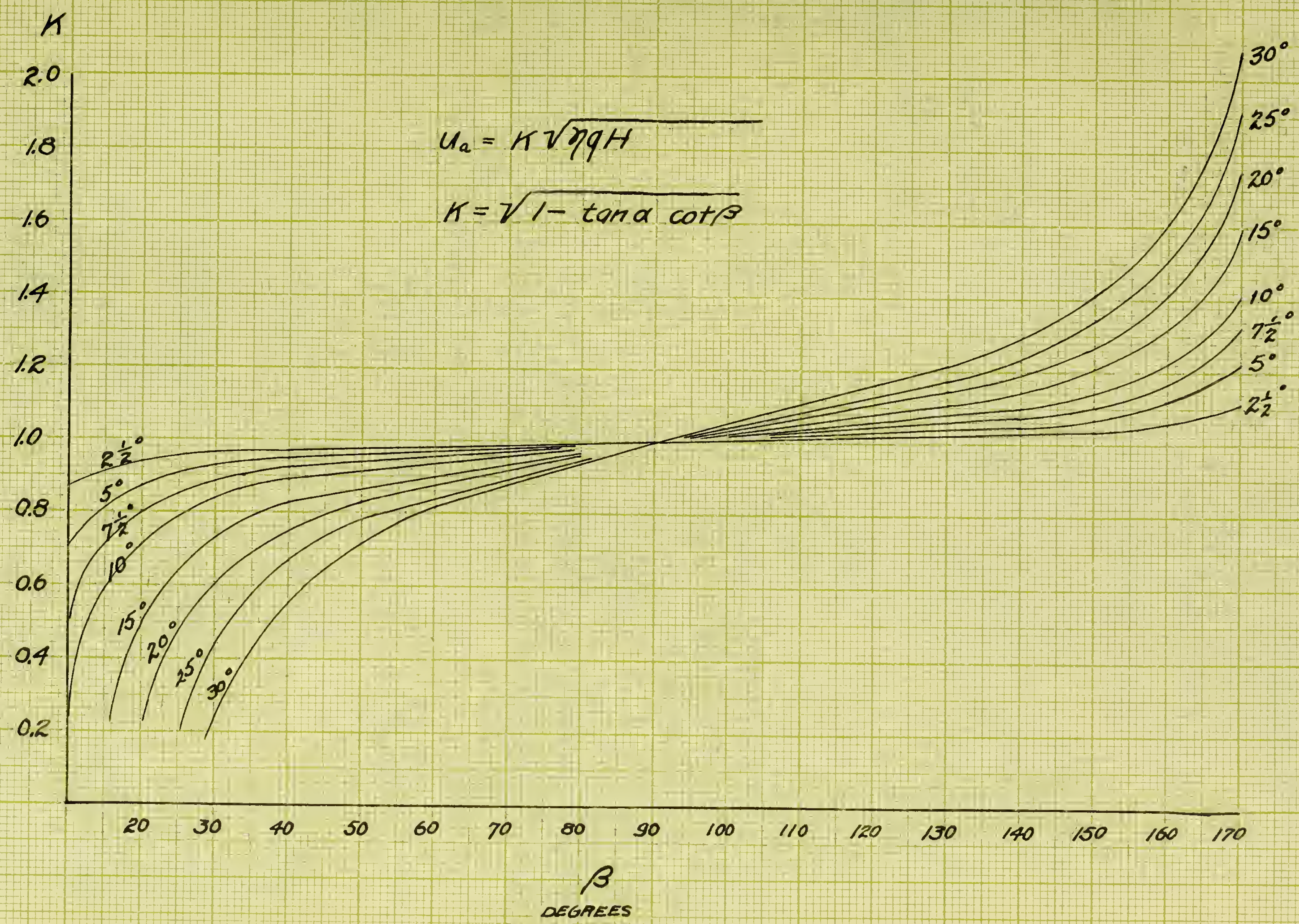
a_e = The perpendicular width of the channel at entrance.

The first step in the design of the impeller is in the choice of the angles β_a and α. For low-lift pumps with small discharge, β_a and α should both be large. β_a should be close to 160° and α should be around 25°. For high lift pumps with small discharge, α should be as small as possible and β_a should be close to 90°, even being less in extreme cases.

For the pump taken up in this thesis, the angles were chosen as follows:

$$\beta_a = 150°$$
$$\alpha = 20°$$

We are now able to obtain, from our fundamental equation values for κ and u_a. For convenience in the work a series of curves was plotted between κ, β, and α. They are shown in the accompanying



EUGENE DIETZGEN CO., CHICAGO.

photograph. From these curves we get:

$$K\ (\text{For } \beta_a = 150° \text{ and } \alpha = 20°) = 1.277$$

$$u_a = K\sqrt{\eta g H} = 1.277\sqrt{1.43 \times 25 \times 32.2} =$$

$$= 1.277 \times 33.9 = 43.25 \text{ ft/sec.}$$

We may now draw our velocity polygon for exit is shown in Fig. IV, from which we obtain;

$$V_a = 19.34 \text{ ft/sec.}$$

$$W_a = 28.3 \text{ ft/sec.}$$

$$V_r = 9.67 \text{ ft/sec.}$$

The next step in the design is in the determination of the diameter D_a, upon which all exit calculations are based.

From Fig. V, it is evident that

$$a_a + S_a = \frac{D_a \pi \sin \beta_a}{Z} \qquad (1)$$

Also: from Fig IX.

$$D_a = \sqrt{(D_a^a)^2 + (a_a)^2 + 2\, a_a D_a \cos \beta_a^a} \qquad (2)$$

The best way to determine

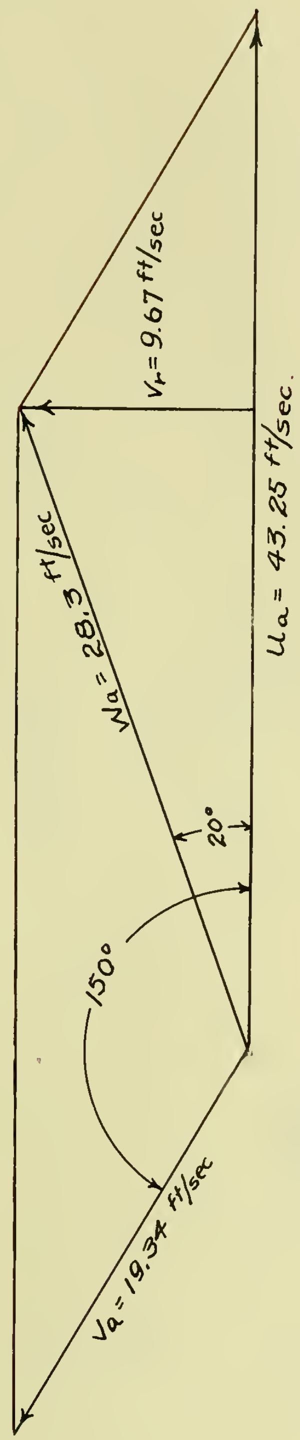


Fig. IV

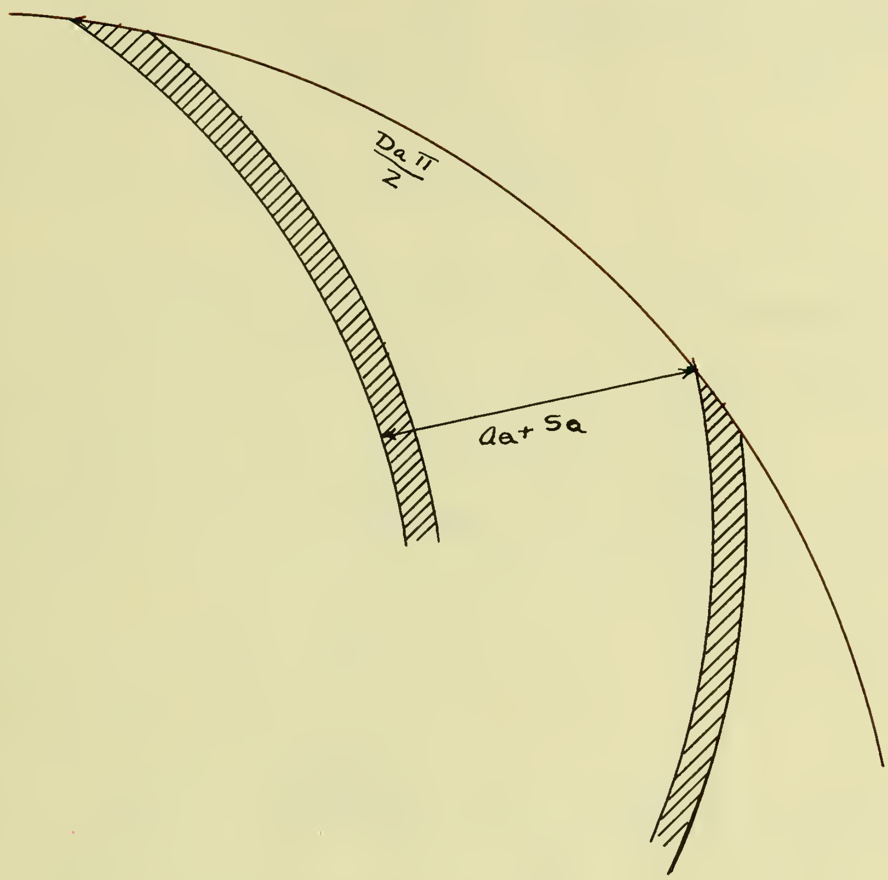


Fig. V.

D_a is by the cut-and-try method. An even value is assumed for D_a^a. Values are then assumed for S_a and D_a and a_a determined from equation (1). The value of $\cos \beta_a^a$ may be found from the relation that.

$$\frac{D_a^a}{D_a} = \frac{\sin \beta_a}{\sin \beta_a^a}$$

By substuting values alread found in equation (2), our assumed value of D_a can be checked.

In this problem, we assumed;

$S_a = .1875''$

$D_a = 8.70''$

$D_a^a = 10.0''$

$$a_a = \frac{8.70 \times \pi \times .5}{8} - .1875 = 1.52''$$

$$D_a = \sqrt{100 + 2.31 - 26.3} = 8.70''$$

We next determine the speed of the pump.

$$R.P.M. = \frac{u_a \times 60}{D_a \pi} = \frac{43.25 \times 60}{\frac{8.7}{12} \times \pi} = 1140$$

The next step is the determination of the thickness of the impeller, b_a, and the area of discharge F_a.

$$Q' = v_r \times \text{area of discharge}$$

Now the area of discharge,

$$= b_a \pi D_a - \text{Area occupied by vanes}$$

$$= b_a \left[\pi D_a - Z S_a \frac{1}{\sin \beta_a} \right]$$

$$= b_a \left[\pi D_a - Z \frac{S_a}{\sin \beta_a} \right]$$

In this determination, b_a was assumed as the even fraction of an inch which would bring the discharge of the pump the nearest to the assumed value. We assumed;

$$b_a = .375''$$

$$Q' = 9.67 \times \frac{1}{144} \left[\pi \times 8.7 - \frac{8 \times .1875}{.5} \right] .375$$

$$= .615 \text{ cu. ft. per sec.}$$

Making allowance for slip which we assume as 6%, we get;

$$Q = \frac{.615}{1.06} \times 7.48 \times 60 = 261 \text{ gal. per. min.}$$

$$F_a = .375\left[\pi D_a - \frac{Z\,S_a}{\sin\beta_a}\right]$$

$$= .375\left[\pi\, 8.7 - \frac{8\times.1875}{.5}\right]$$

$$= 9.15 \text{ sq. in.}$$

We now start on our computations at entrance. The first thing is the determination of W_s', the velocity of suction. We will assume the impeller cross-section as in Fig. VI.

The net area of suction is then;

$$\frac{\pi}{4}(16.0-1.56) = 11.35 \text{ sq. in.}$$

$$W_s' = \frac{Q'}{11.35} = \frac{.615\times144}{11.35} = 7.80 \text{ ft/sec.}$$

The next thing is the assumption of some value for De, which is usually taken as a little larger or smaller than the diameter of suction. In this case it was taken as 4.5."

From this we obtain;

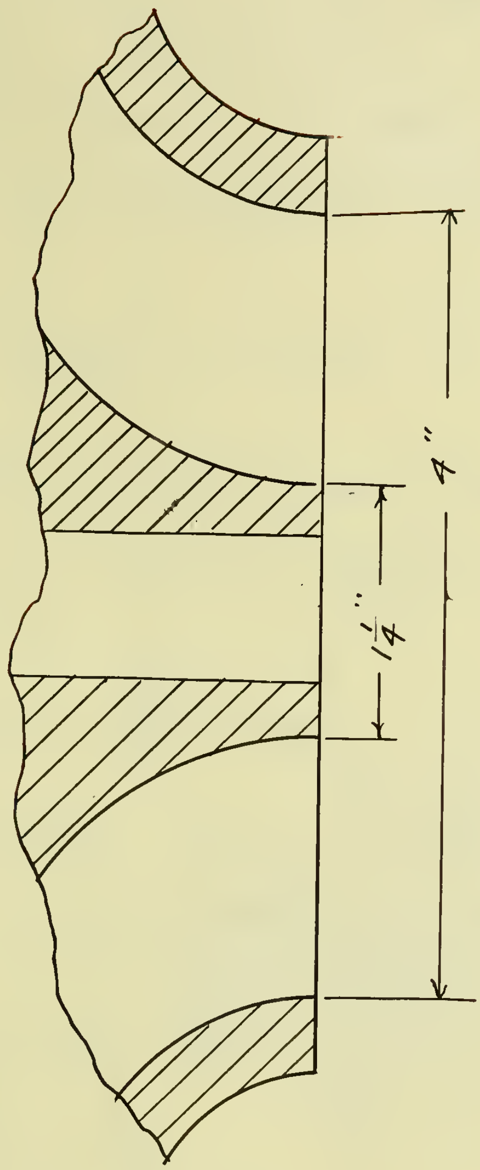


Fig. VI.

$$u_e = \frac{4.5}{8.7} \times 43.25 = 22.4 \text{ ft/sec.}$$

We can also get t_e, which is the distance along the circumference of the circle of diameter D_e, occupied by one channel.

$$t_e = \frac{\pi \times 4.5}{8} = 1.77''$$

At entrance, as at exit, we have the approximate relation that;

$$\frac{a_e + S_e}{t_e} = \sin \beta_e \qquad (3)$$

Also, since W_e must be radial, or in other words $90°$

$$V_e = \sqrt{u_e^2 + W_e^2} \qquad (4)$$

$$\sin \beta_e = \frac{W_e}{V_e} \qquad (5)$$

$$W_e = W_s' \frac{a_e + S_e}{a_e} \qquad (6)$$

From the relations (3) and (5), we may solve for a_e and β_e by the cut-and-try

method, or the graphical construction shown in Fig. VII may be used.

AB was laid off equal to $u_e = 22.4$ ft/sec.

AE was laid off equal to $W_s' = 7.8$ ft/sec.

EF was laid off equal to $s_e = .1875''$

Through F a line FG was drawn parallel to B.E.

Arc BG of radius t_e was struck intersecting FG at G.

GM was drawn intersecting AB at M. GM is perpendicular to AB.

BG was drawn intersecting AE at C.

Relations (3), (4), (5) and (6) are all satisfied by the figure, for:

$$BC = V_e = 24.7 \text{ ft/sec.}$$

$$AC = W_e = 10.4 \text{ ft/sec.}$$

$$MG = (a_e + s_e) = .75''$$

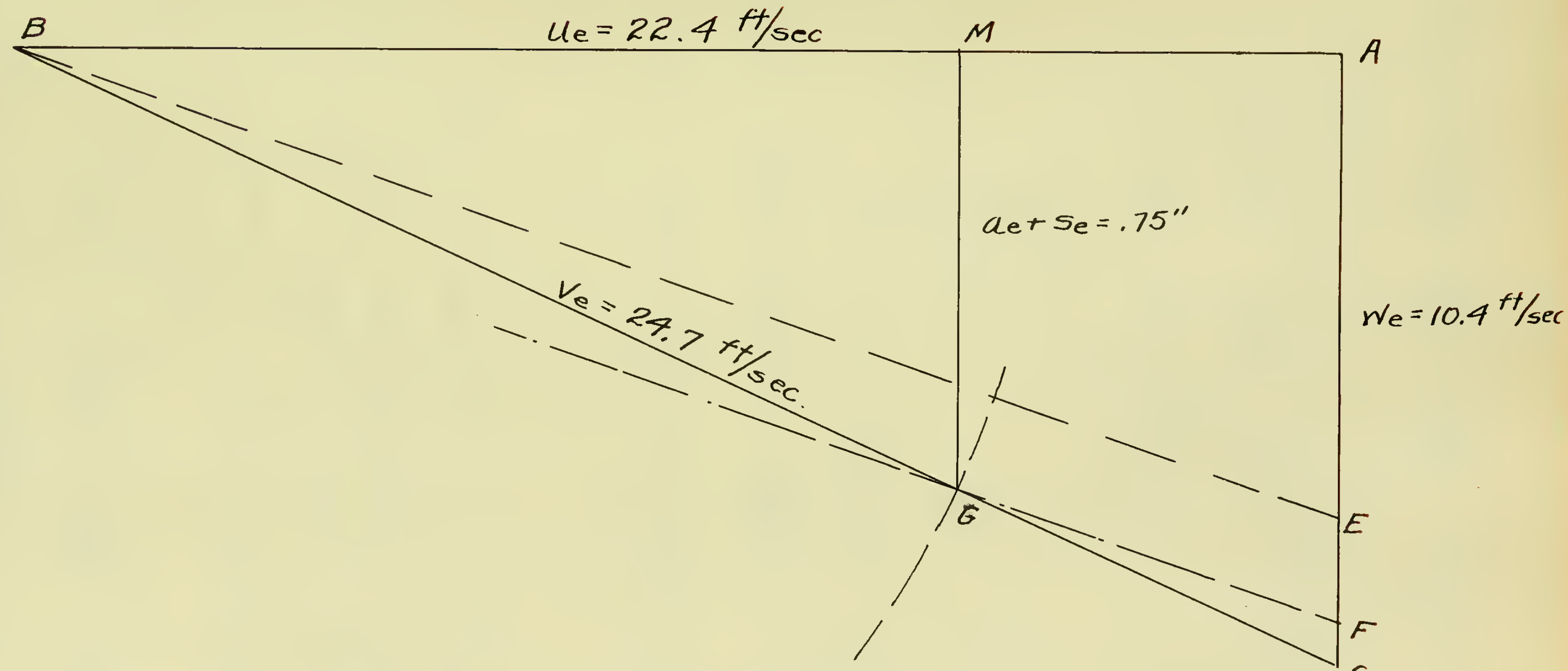


Fig. VII.

$$\sin\beta_e = \frac{AC}{BC} = \frac{W_e}{V_e} = \frac{MG}{BG} = \frac{a_e + S_e}{t_e} = .422$$

The area of entrance,

F_e, $= \left(\pi D_e - \frac{Z S_e}{\sin\beta_e}\right) b_e$

$$= \left(\pi\, 4.5 - \frac{8 \times .1875}{.422}\right) b_e$$

$$= 10.55\ b_e$$

Now,

$$Q' = W_e F_e$$

From which,

$$b_e = \frac{.615 \times 144}{10.55 \times 10.4} = .808''$$

$$F_e = .808 \times 10.55 = 8.52 \text{ sq.in.}$$

Now that we have determined our velocities at both entrance and exit, we may check our work by means of the relation that;

$$\eta H = \frac{U_a^2 - U_e^2}{2g} + \frac{V_e^2 - V_a^2}{2g} + \frac{W_a^2}{2g} - \frac{W_e^2}{2g} \qquad (7)$$

which says, that ηH must equal: the head due to the change in

peripherial velocity or $\frac{u_a^2 - u_e^2}{2g}$; plus the head due to the change in relative velocity or $\frac{v_e^2 - v_a^2}{2g}$; plus the head due to the absolute velocity of exit or $\frac{W_a^2}{2g}$; minus the head due to the absolute velocity of entrance or $\frac{W_e^2}{2g}$.

Substituting the values we have already obtained, in equation (6) we get;

$$\eta H = \frac{(43.25)^2 - (22.4)^2}{64.4} + \frac{(24.7)^2 - (19.34)^2}{64.4} + \frac{(28.3)^2}{64.4} - \frac{(10.4)^2}{64.4}$$

$$= \frac{2295}{64.4} = 35.8 \text{ ft.}$$

$$\eta = \frac{35.8}{25} = 1.43$$

Our next step is to obtain the charachteristic equation of the pump, which is the equation showing the relation between head and discharge.

From the two velocity polygons we have the relations that

$$v_a = \frac{v_r}{\sin \beta_a} \quad \text{and} \quad W_e = v_e \sin \beta_e$$

Now,

$$V_r = \frac{F_e}{F_a} W_e$$

From which relations, we obtain

$$\frac{V_e^2 - V_a^2}{2g} = \left[\frac{V_r^2}{\sin^2\beta_e}\,\frac{F_a^2}{F_e^2}\right] - \left[\frac{V_r^2}{\sin^2\beta_e}\right] \qquad (8)$$

From the polygon at exit, we have;

$$\frac{W_a^2}{2g} = \frac{\left(U_a + \frac{V_r}{\tan\beta_a}\right)^2 + V_r^2}{2g} \qquad (9)$$

Likewise, from the polygon at entrance;

$$W_e^2 = \left[V_r^2 \frac{F_a^2}{F_e^2}\right] + \left[U_e - \frac{V_r}{\tan\beta_e}\,\frac{F_a}{F_e}\right]^2 \qquad (10)$$

Substituting (8), (9), and (10) in (7) and simplifying, we have;

$$\eta H = \frac{U_a^2 - U_e^2}{2g} + \frac{U_a^2 - U_e^2}{2g} + 2V_r \frac{\left(\frac{U_a}{\tan\beta_a} + \frac{U_e}{\tan\beta_e} \times \frac{F_a}{F_e}\right)}{2g} \qquad (11)$$

Let $\frac{U_a^2 - U_e^2}{g} = C_1$

Also let $\frac{\left(\frac{U_a^2}{\tan\beta_a} + \frac{U_e}{\tan\beta_e} \times \frac{F_a}{F_e}\right)}{g} = C_2$

Then we get:

$$\eta H = C_1 + C_2 V_r$$

which is the charachteristic equation of the pump. When plotted it takes the form shown in Fig. VIII.

$$c_1 = 42.5 \qquad c_2 = -.726$$

$$\eta H = 42.5 - 7.02 = 35.5 \text{ ft.}$$

$$\eta = \frac{35.5}{25} = 1.425$$

The calculations are now completed and checked, and the next step is in the construction of the vanes themselves.

Since the direction of the relative velocities must be perpendicular to the channel cross-section, and also the distances a_e and a_a must be measured perpendicular to the sides, some form of blade must be taken which does this. The involute of a circle fulfills these conditions and is

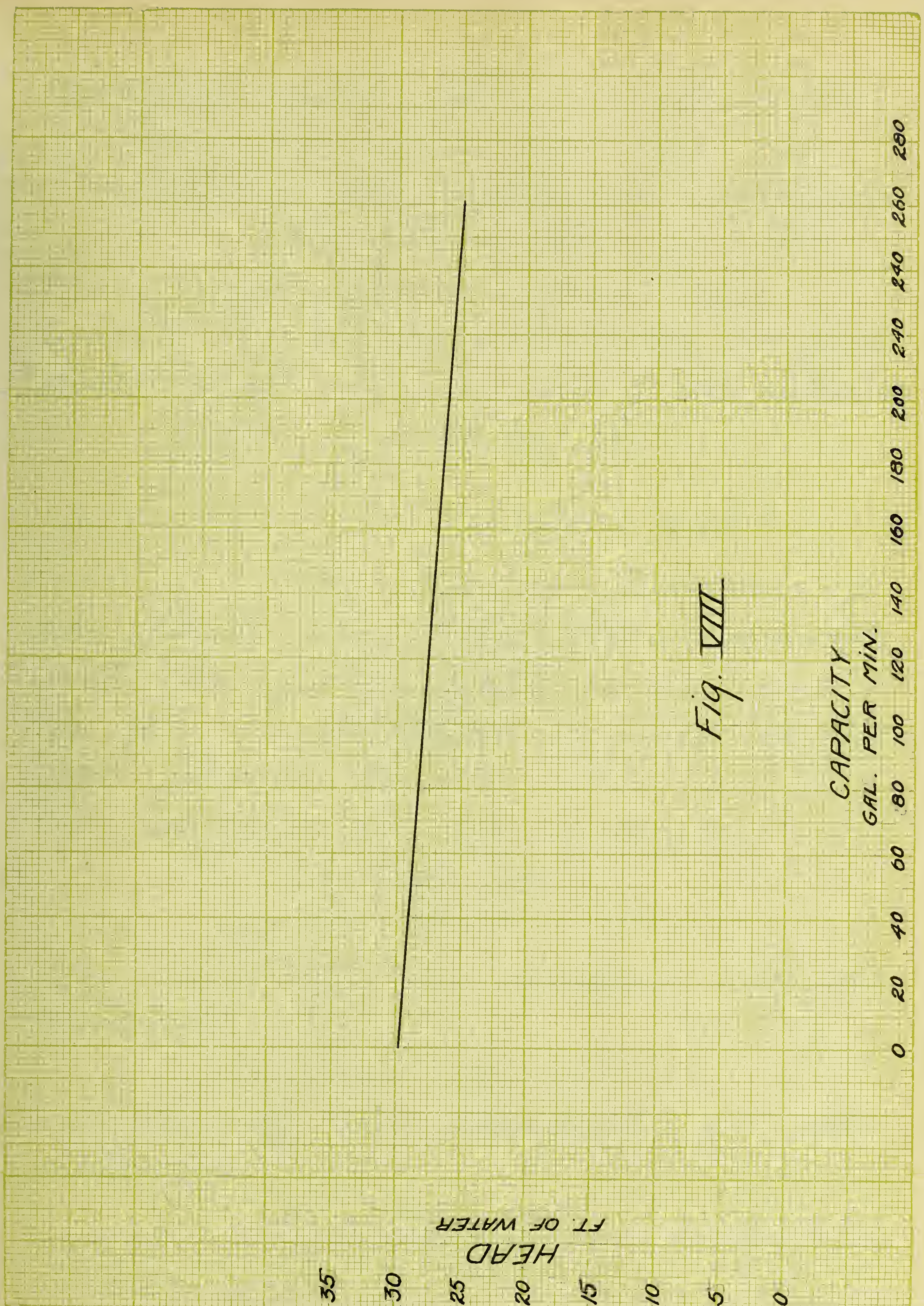


Fig. VIII

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therefore, used. The construction of this involute at exit is made in the following manner.

The diameter of the base circle upon which the involute is constructed is first determined by means of the relation,

$$d_a = \frac{Z(a_a + s_a)}{\pi}$$

$$= \frac{8 \times 1.71}{\pi} = 4.35''$$

The circumference of the circle D_a was then divided into $2Z$ parts as shown in Fig. IX, and these points tangents were drawn to the base circle. On tangent AB the distances

$$CN = \frac{a_a}{2} = .76''$$

$$\text{and } BN = \frac{a_a}{2} + s_a = .95''$$

were laid off. On tangent HE, the distances

$$DF = DE = \frac{s_a}{2} = .094''$$

were laid off. On tangent JK, the distances

$$KL = \frac{a_a}{2} = .76''$$

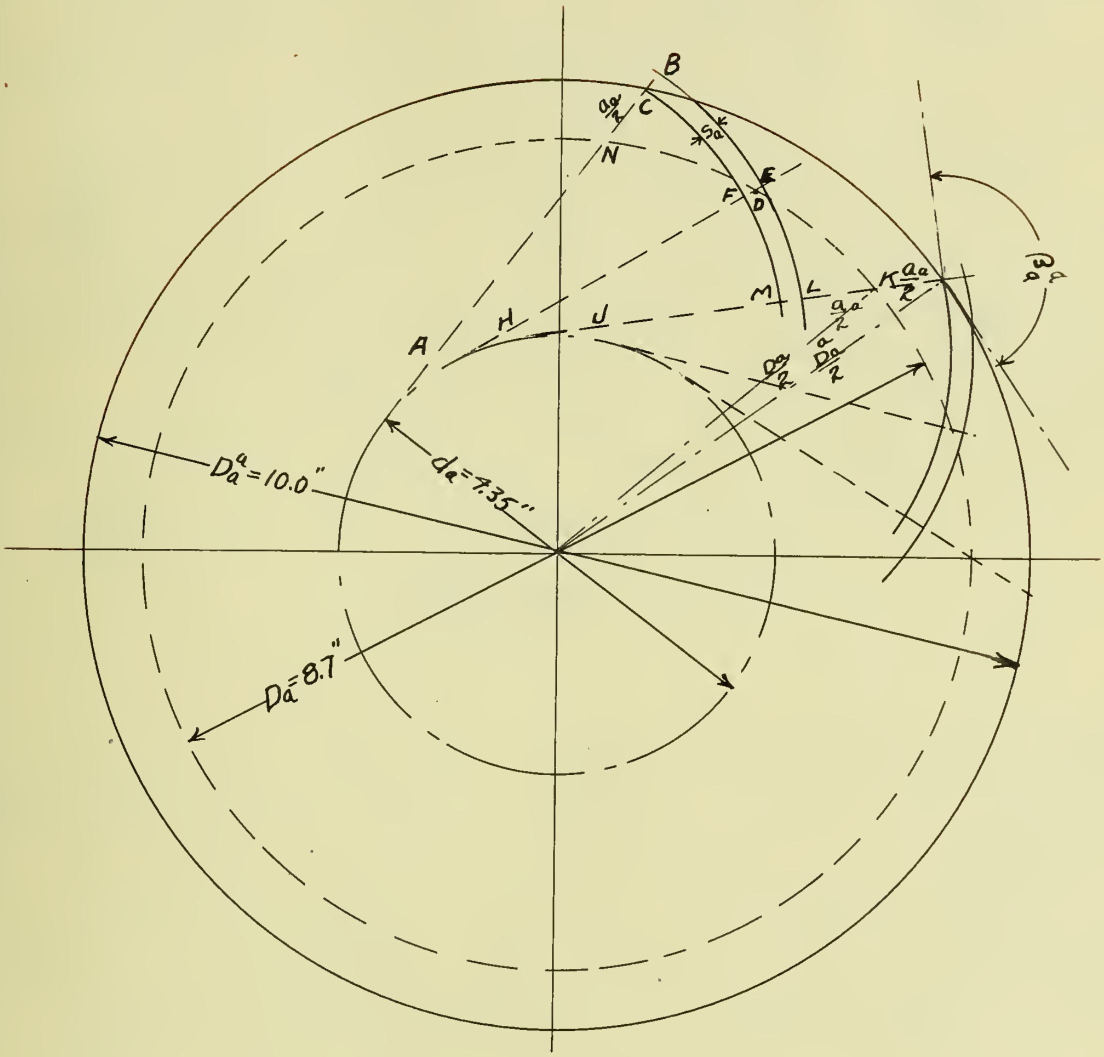


Fig. IX.

and $KM = \frac{4a}{2} + 5a = .95''$

were laid off. This gives us three points on each edge of the blade cross-section. The involute was then replaced by the arc of a circle having its center in the base circle and passing through the three points.

The same construction was used at entrance as shown by Fig. X. In this case

$$d_e = \frac{8 \times .75}{\pi} = 1.91''$$

After the involutes have been constructed they are connected by a straight line or smooth curve, so as to give a channel of approximately constantly increasing width. In this case the arc of a circle was used.

The next step after the blades have been drawn is

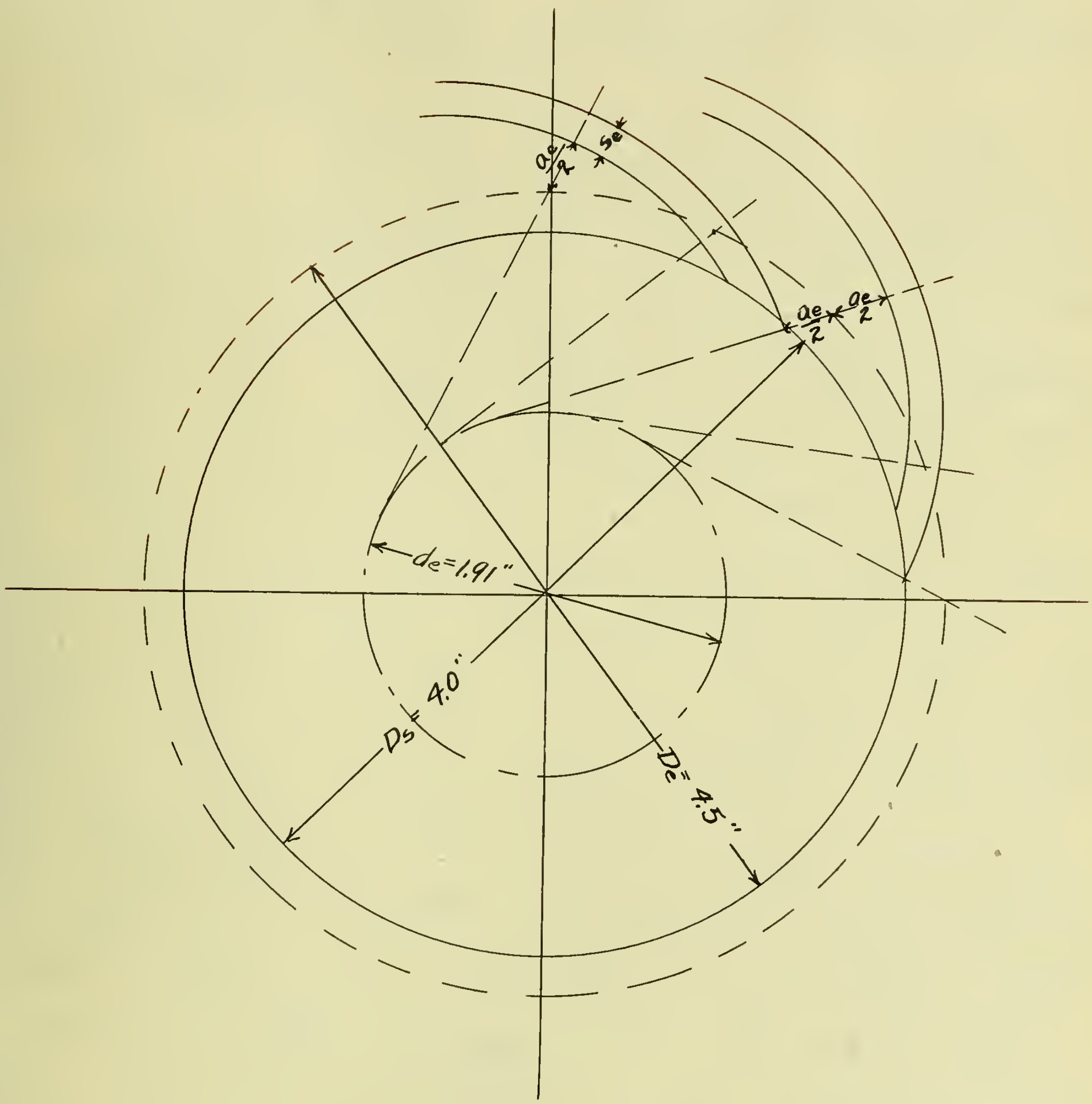


Fig. X.

the determination of the width of the impeller at different diameters. The method used was as follows.

At any point the relative velocity v, multiplied by the area of the passages must equal the discharge of the pump or Q'. In order to find the width of the channel, circles were drawn in the passage as shown in Fig. XI. The points of tangency with the blades were then connected by a line and the line bisected. The length of this line represented the width of the channel at that point, and its center marks the center of the channel. Enough of these central points were found to enable the curve marking the center of the passage to be laid out on a horizontal axis. At diameters $D_a = 8.1''$ and $D_e = 4.5''$, the velocites $v_a = 19.34$ and $v_e = 24.7$ were

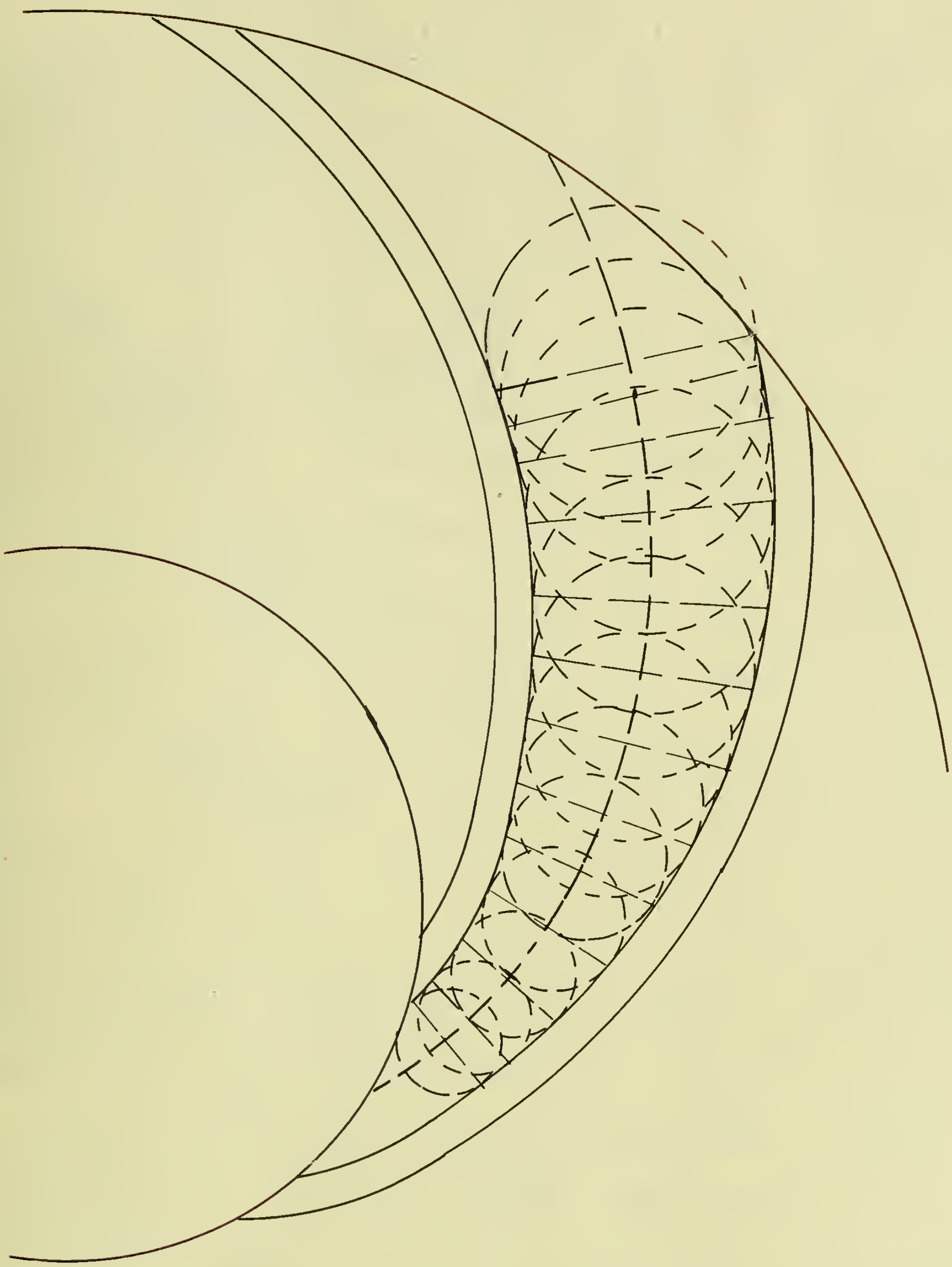


Fig. XI.

known. These values were then scaled off as ordinates at the corresponding points of the developed center line. A straight line relation was assumed between these velocities as shown in Fig. XII and the velocities for the points at the different diameters found. Having the relative velocities and the channel widths the thickness of the impeller was easily calculated, the results being shown in Table I.

The suction channel must be of constant area, and, since the areas must be measured to a great extent on the frustrum of a cone, a cut and try method is the only one practicable. The boundaries must be drawn in and the passage checked for constant area.

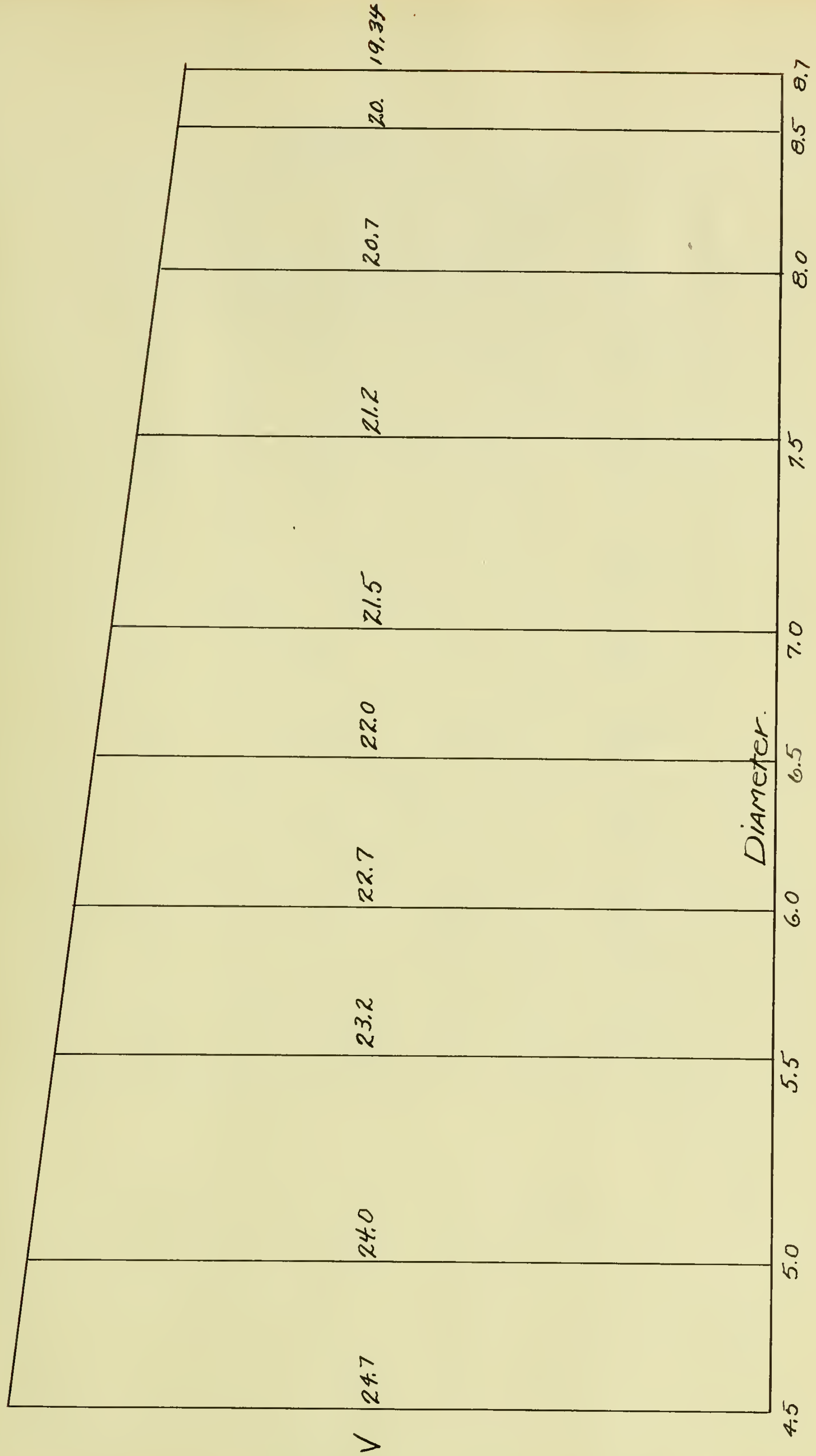

V
24.7
24.0
23.2
22.7
22.0
21.5
21.2
20.7
20.
19.34
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
8.7
Diameter.

Fig. XII.

TABLE I.

Diameter	V ft. per sec.	Width in.	b in.
4.5	24.7	0.56	.808
5.0	24.0	0.70	.610
5.5	23.2	0.84	.570
6.0	22.7	1.00	.488
6.5	22.0	1.17	.431
7.0	21.5	1.30	.398
7.5	21.2	1.33	.396
8.0	20.7	1.40	.384
8.5	20.0	1.46	.380
8.7	19.34	1.52	.375

Chapter III.
Design of Casing.

The only theory connected design of the discharge is the self-evident hat; beginning with of maximum discharge sectional area must decrease, and this must be proportional gle through which moved. The areas and, cross-section was taken the corresponding diameters in Table II.

Chapter III.
Design of Casing.

The only theory connected with the design of the discharge chamber, is the self-evident relation that; beginning with the point of maximum discharge the cross-sectional area must constantly decrease, and this decrease must be proportional to the angle through which we have moved. The areas and, since the cross-section was taken as circular, the corresponding diameters are given in Table II.

TABLE II.

ANGLE	SECTION AREA sq. in.	DIAMETER in.
0°	7.07	3.0"
45°	6.18	2.81"
90°	5.30	2.60"
135°	4.42	2.37"
180°	3.54	2.12"
225°	2.64	1.83"
270°	1.77	1.50"
315°	0.88	1.06"

Chapter IV.

Design of Shaft.

The shaft at the impeller is acted upon by torsion alone, since the water pressure is everywhere equally distributed. To find the magnitude of this torsion we first find the horse-power required to drive the pump. This horse-power will equal, the product of the discharge, the specific weight of water, and the head, divided by 550 times the efficiency. Or

$$H.P. = \frac{.58 \times 62.5 \times 25}{.7 \times 550} = 2.33.$$

The torque then

$$= \frac{2.33 \times 33000 \times 12}{1140\ \pi} = 268 \text{ in. lb.}$$

Assuming a $\frac{3}{4}$" shaft and checking for the stress, we

obtain;

$$S = \frac{268\,\pi}{.422 \times 16} = 3,175 \text{ lb. per in.}$$

Chapter V.
Details.

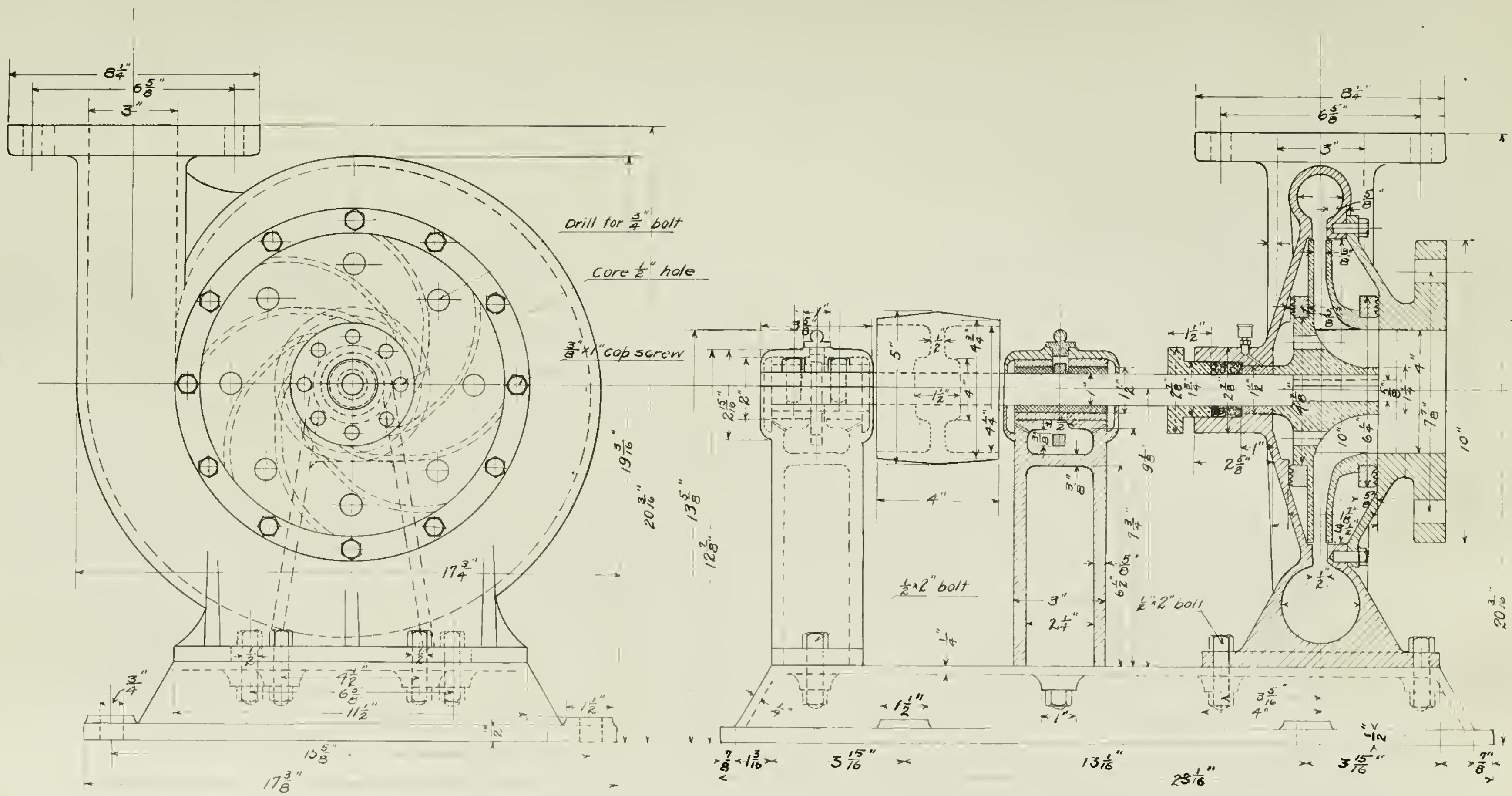
The remainder of the design of the pump is almost wholly a matter of judgement with the designer. Most of the work is entirely empirical, but whenever possible calculations should be made to check the work.

One thing that is absolutely necessary is a stuffing box. Unless the shaft is provided with one a leakage of air is bound to result, impairing the efficiency of the pump or even making it impossible to start. Space for plenty of packing should be provided and sufficient clearance for packing also.

Another detail which adds markedly to the efficiency of the pump is the use of labyrinth rings. They may be of several forms, the tongue-and-groove type being the most common. In this pump they are merely a series projections of tooth-like cross-section running against faced surfaces on the casing.

In thickness of metal, sizes of bolts et cetera, common practise and the designer's judgement must hold as all calculated sizes would be entirely too small.

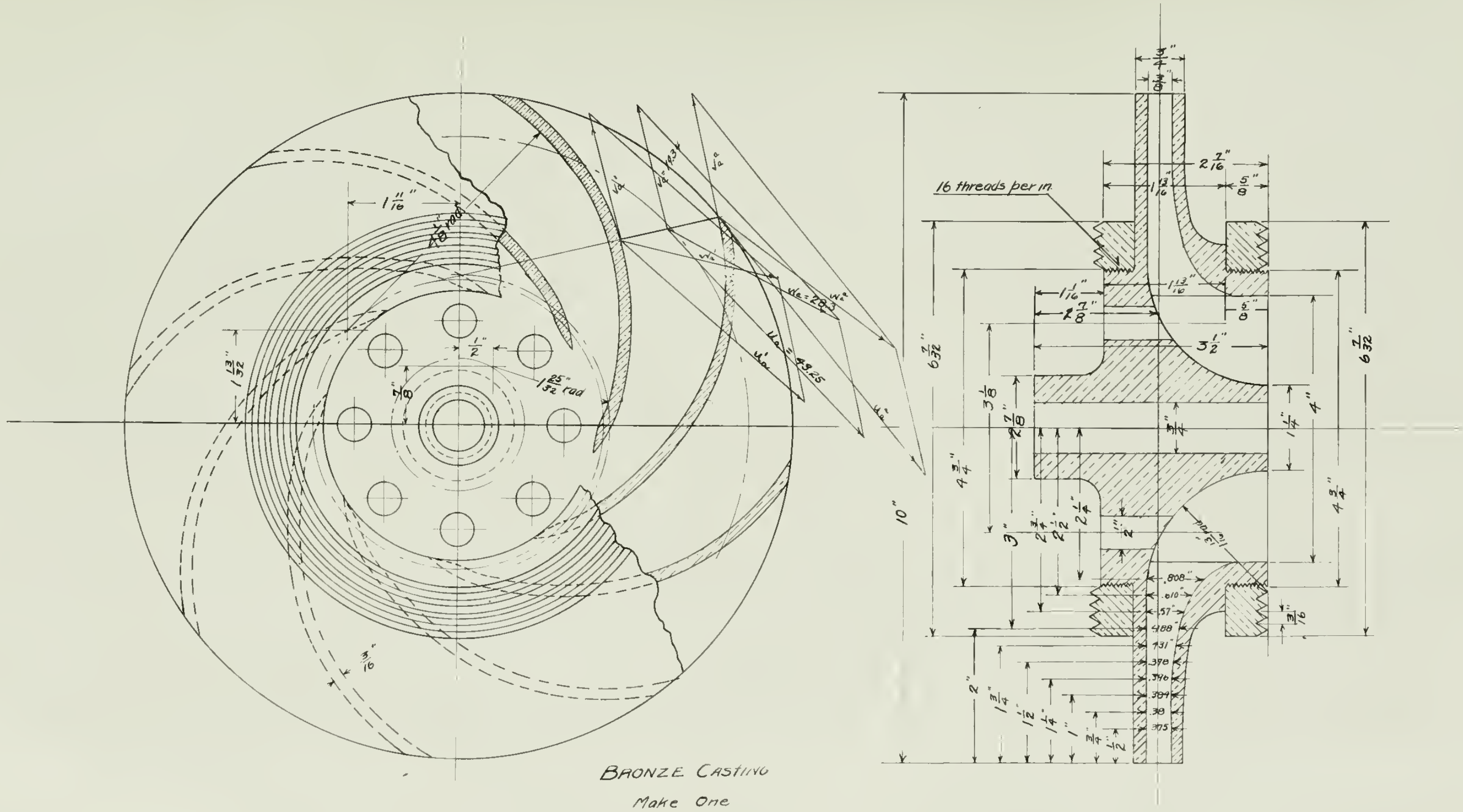
ASSEMBLY VIEW



3 IN. CENTRIFUGAL PUMP

260 GAL. PER MIN.   25 FT. HEAD

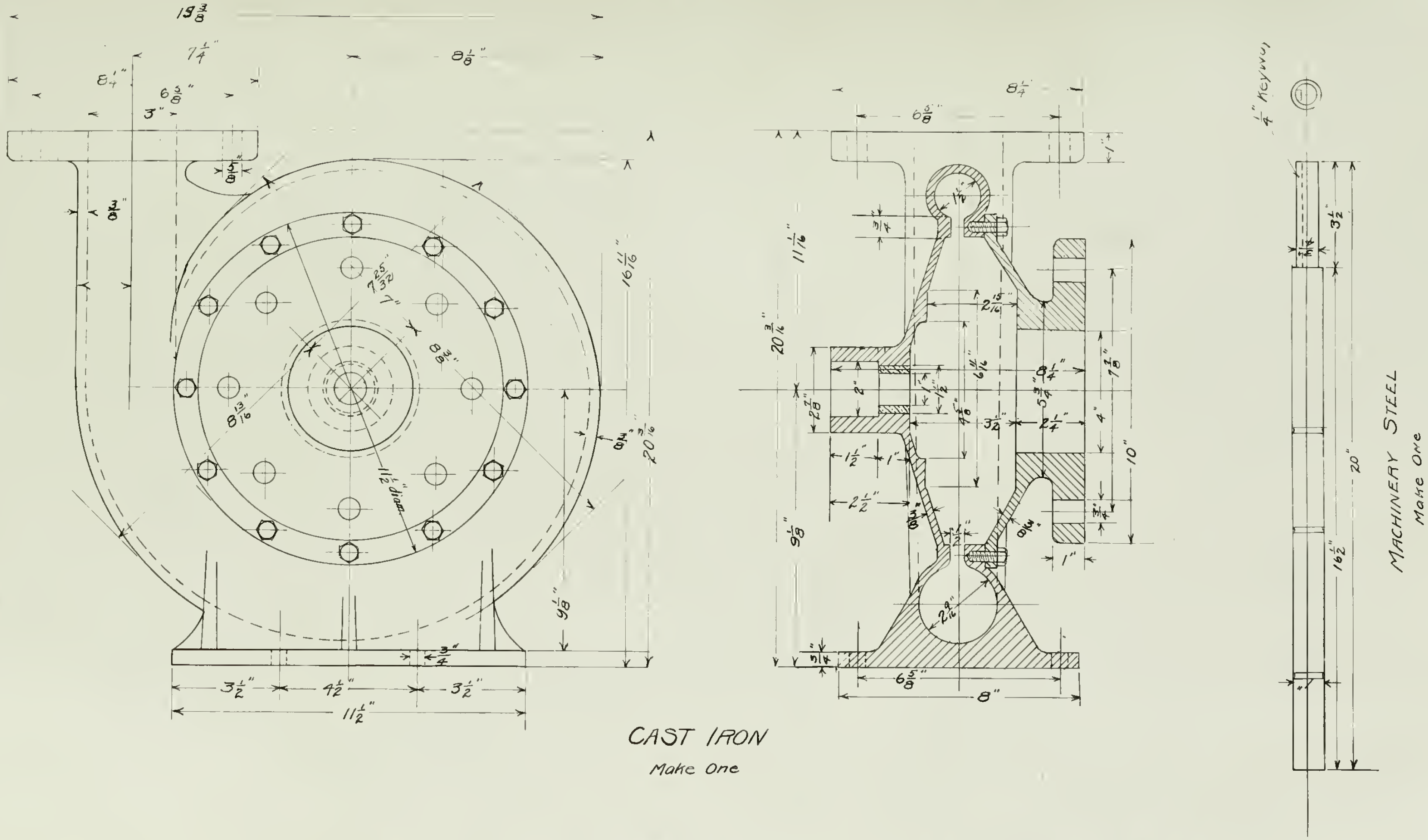
DETAIL OF IMPELLER



3 IN. CENTRIFUGAL PUMP

260 GAL. PER MIN. 25 FT. HEAD.

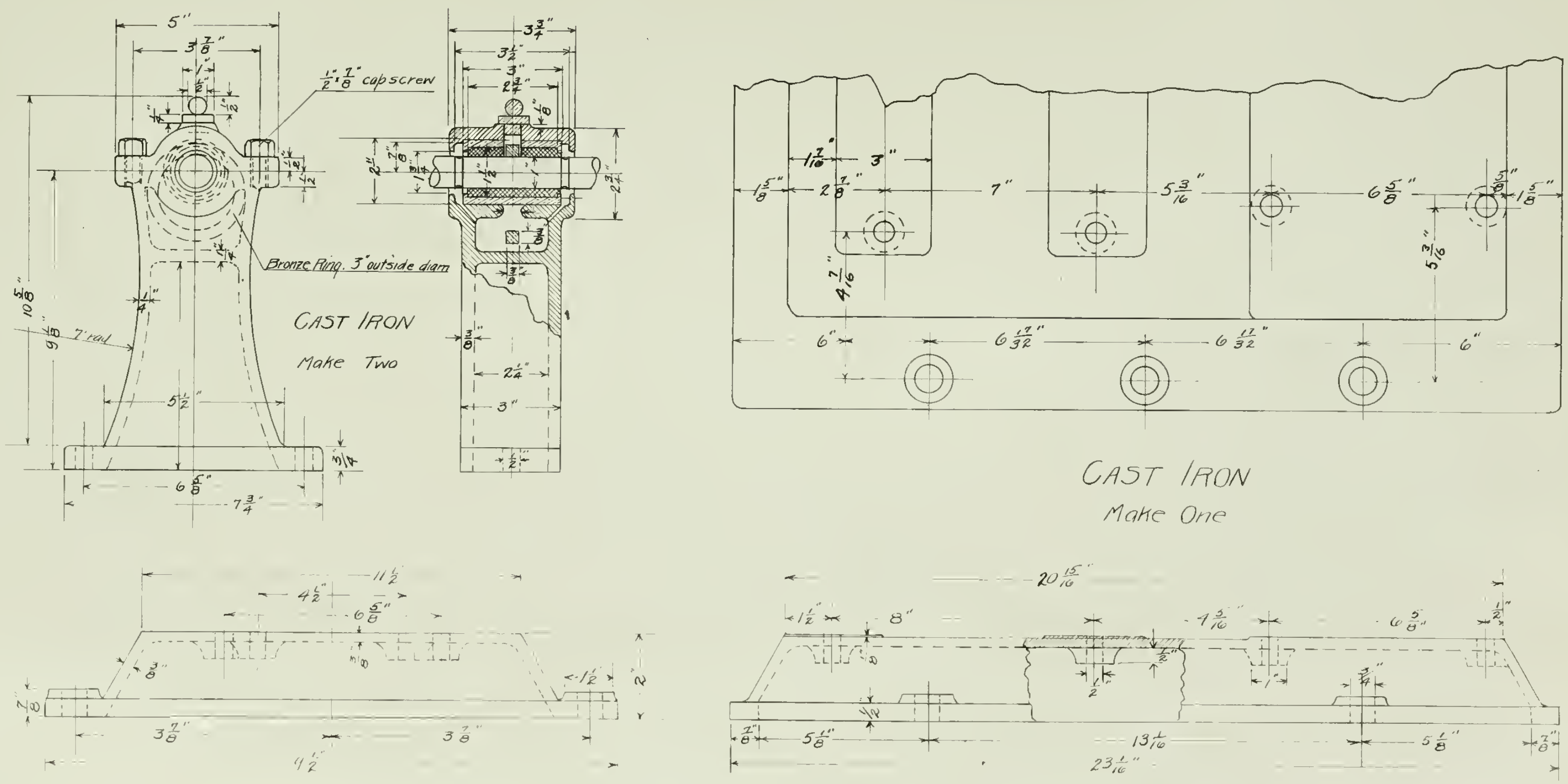
DETAIL OF CASING AND SHAFT



3IN. CENTRIFUGAL PUMP

260 GAL. PER MIN. 25 FT. HEAD

DETAIL OF BEARING AND STAND



3IN. CENTRIFUGAL PUMP

260 GAL. PER MIN. 25 FT. HEAD

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